NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

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COMPARATIVE EFFECTIVENESS OF A CONVECTION-TYPE AND A

RADIATION-TYPE COOLING CAP ON A TURBOSUPERCHARGER

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SUMMARY

An investigation was made to compare the effectiveness of two basic methods of cooling a turbosupercharger and to obtain data on the temperature level and distribution in the turbine wheel. Turbine-wheel temperatures were obtained for various conditions of cooling-air mass flow, exhaust-gas temperature, and exhaust-gas mass flow. Modi-fied forms of the standard cooling caps were used to allow for the installation of temperature-measuring devices.

The convection-type cap that employs a jet of air blowing on the rim of the wheel gave temperature reductions of the order of 150° F when the air flow was increased from 0 to 3 pounds per minute (average flow for a flight installation), and further temperature reductions were attained with increased air flow. With the radiation-type cap in which the cooling air never comes in contact with the turbine wheel, maximum reductions in wheel temperature of 20° F to 30° F were attained with air flows up to 3 pounds per minute and no additional cooling was obtained with greater flows.

INTRODUCTION

The efficiency and power of a turbine can be increased with increases in gas temperature. The operating gas temperature being limited by the capacity of the blade and wheel material to withstand centrifugal and other stresses at high temperatures, cooling of the wheel will permit the use of a higher gas temperature or of a lighter wheel for the same gas temperature, stress, and life.

Two basic methods of cooling are used in cooling caps for turbine wheels of turbosuperchargers. One cap employs a jet or jets of cooling air directed on the rim of the wheel just inside the blade ring and is commonly known as a convection cooling cap. The other type, called a radiation cooling cap, has a cool surface close to the wheel to which the heat from the wheel may radiate.

Although it is known from tests conducted by the General Electric Company that the radiation cap has a lesser cooling effect than the

convection cap, other factors influence the selection of one cap or the other. Under certain conditions of engine operation involving high fuel-air ratios and high exhaust-gas temperatures, air introduced into the exhaust system of a turbosupercharger will combine with unburned fuel and cause afterburning, which results in excessive temperatures and may cause turbine-wheel failure. With the convection cap, the possibility of afterburning is always present; whereas, with the radiation cap, the cooling air is discharged into the slipstream at some point outside the exhaust hood.

Another consideration is the design of the exhaust hood, which so directs the exhaust gases from the turbine that they enter the slipstream smoothly. The turning of the gases gives rise to an unequal pressure distribution in the hood close to the wheel and tends to cause a flow of gas across the face of the wheel under the cooling cap. This cross flow of gas, which affects the temperature level of the wheel, may be minimized by variation in the shape of the surface of the radiation cooling cap close to the wheel.

The determination of temperatures on a turbine wheel spinning at high speed presents a difficult problem. The principal methods available are: (1) chemical paints that change color with temperature (references 1 and 2), (2) fusible plugs inserted in the wheel that melt at different temperatures (reference 1), (3) radiation pyrometers (references 1 and 3), (4) electromotive force from thermocouples on the surface of the wheel taken off by induction (reference 4), and (5) electromotive force from thermocouples on the surface of the wheel taken off with slip rings (references 1 and 5).

A device for measuring turbine-wheel temperatures should be easily installed, compact, accurate, and free from mechanical difficulties. Chemical paints or fusible plugs are difficult to apply as the test rig must be partly dismantled to recoat the surface or to insert new plugs after each test-point determination. This procedure is unsatisfactory for establishing a curve of temperature against a given operating condition. The radiation pyrometer requires very careful control of the temperature of the pyrometer casing and is subject to errors due to variations in the emissivity of the wheel surface. Furthermore, the necessity of a clear viewing space interferes with the installation of the cooling cap. The induction pickup must be shielded to prevent error introduced by the earth's magnetic field and is too bulky and too delicate for the present application.

As part of a general investigation being conducted by the NACA to improve the performance of superchargers and turbines, the Cleveland laboratory conducted an investigation to evaluate quantitatively the comparative disk cooling obtained with these two basic methods of cooling and to obtain data on the actual temperature level and distribution in the turbine wheel under controlled operating conditions.

The exhaust hood was made symmetrical in order to eliminate the variables of hood design. The temperature was measured by thermocouples with a slip-ring pickup, the rings and contacts of which were designed and developed by Alfred W. Young and Edward R. Bartoo of the NACA staff.

Curves are presented of turbine-wheel temperatures plotted against cooling-cap air flow for both methods of cooling, against exhaust-gas temperature, against exhaust-gas mass flow, and against time to reach equilibrium conditions.

APPARATUS

A schematic diagram of the test equipment is shown in figure 1. A liquid-cooled aircraft engine was used to supply exhaust gas to the turbosupercharger, and the engine power was absorbed by a water brake. The exhaust-gas flow could be varied by varying the engine power; and the exhaust-gas temperature, by changing the fuel-air ratio. The engine exhaust was led to an altitude chamber, and a waste pipe was used for bypassing the flow during starting and stopping. This waste pipe was equipped with a gate valve rather than with the loosely fitting butterfly valve normally used with turbosupercharger installations. During testing, this valve was kept tightly closed in order that all the measured gas from the engine would pass through the turbine. The altitude chamber was connected by means of a large duct to the laboratory altitude system and the pressure in the chamber could be maintained at any value from atmospheric pressure to 4 inches of mercury absolute.

A production turbosupercharger was installed inside the altitude chamber as shown by dotted lines in figure 1. Details of the turbine are shown in figure 2. The cowlings, the exhaust hood E, the slip-ring pickup, the cooling cap D, and seal C are not standard equipment but were added for these tests. The purpose of seal C was to prevent mixing of the bearing cooling air and the exhaust gas between the nozzles and the blades. The exhaust hood was made symmetrical to minimize flow of gas across the face of the wheel. The hood was water-cooled to eliminate excessive heat in the space where the thermocouple pickup was located.

The cooling caps devised for these tests are shown in figure 3 and are representative of the two basic cooling methods used in production caps. The radiation-type cap consisted of a simple annular ring divided into three passages by internal fins. Air entered through a duct at one side and passed through the annular channels and out at the other side. For the convection-type cap, slots were cut in

3

the bottom of the radiation-type cap, as indicated in figure 3, and the cooling-air exit was blocked off. The slots were made the same size and shape and at the same radius as on the standard convection cap used on production turbosuperchargers.

A clearance of 0.06 to 0.12 inch is allowed between the wheel and the cooling cap on conventional installations. A clearance of 0.16 inch was used for these tests because the thermocouple clamps projected 0.06 inch from the face of the wheel.

The cooling air for the cooling cap, the bearing housing, and the nozzle box passed through ducts from outside the altitude chamber. The differences between the atmospheric pressure outside the chamber and the altitude pressure inside the chamber forced the cooling air through the ducts, and gate valves in these ducts regulated the flow.

For the slip-ring thermocouple pickup, a set of rings and contacts was designed at the NACA, and preliminary tests indicated that an accuracy of $\pm 25^{\circ}$ F could be expected at 21,000 rpm and greater accuracy, at lower speeds.

The slip rings were mounted on a stub shaft attached to the turbine wheel on the side away from the nozzle box. Four chromel-alumel thermocouples were welded to the wheel at different locations as shown in figure 4. The wires passed through stainless-steel tubes clamped to the wheel surface, along slots in the stub shaft, and into the alternate annular rings of chromel and alumel. A set of chromel and alumel buttons mounted on spring-steel strips in a block of insulating plastic material made contact with the rings. Chromel and alumel wires were run from these buttons to the cold-junction terminals outside the altitude chamber. The block holding the contact buttons was so arranged that it could be moved by a flexible cable extending to the outside of the altitude chamber and contacts were made only when readings were being taken. Mechanical and electrical difficulties were experienced with thermocouple 1 and no successful data were obtained from it. Thermocouple 1 was located three thirty-seconds inch from the edge of the disk. The radial location of the thermocouples was as follows:

Thermocouple	Radius (in.)
1	$4\frac{11}{16}$
2	$4\frac{17}{32}$
3	$4\frac{5}{32}$
4	$2\frac{23}{32}$

4

A manually operated potentiometer, together with a light-beam galvanometer, was used to indicate the temperatures. The temperature of the exhaust gas was measured approximately 5 inches ahead of the nozzle box with a quadruple-shielded thermocouple.

The air weight flow to the engine was determined by means of a thin-plate orifice, and the pressure drop across the orifice was measured with an NACA micromanometer. The gasoline weight flow was measured with a calibrated rotameter. The sum of the gasoline and the air weight flows gave the total exhaust-gas weight flow through the turbine. The cooling-air weight flow to the cooling cap was measured by means of a thin-plate orifice, and the pressure drop across the orifice was measured with a water manometer.

The turbine speed was measured with a standard electric tachometer; the generator was driven from a worm gear on the turbine shaft. Inlet and outlet pressures were measured with mercury manometers connected to static-pressure taps before the nozzle box and in the exhaust hood. Altitude pressure was measured in two places in the altitude chamber with mercury manometers.

TESTS

The engine was started with the waste gate open to permit the exhaust gas to bypass the turbine. As soon as the engine was running smoothly, the waste gate was closed and all the exhaust gas passed through the turbine. The engine was idled until the coolant and the oil reached the proper temperature and then was gradually brought up to power until the desired mass flow and temperature of the exhaust gas were reached. In order to determine the length of time necessary for equilibrium temperatures to be reached, a run was made with all conditions constant and readings were taken continuously from the time exhaust gas was first admitted to the turbine. Figure 5 gives the results of this run with exhaust-gas temperatures and turbine-wheel temperatures plotted against time. These curves indicate that temperature readings should not be taken until at least 20 minutes have elapsed after a change in running conditions that might give rise to a small change in turbine-wheel temperature and at least 1 hour after any change involving a large variation in wheel temperature. All the data presented in this report were taken after 1 hour or more of running with no change in operating conditions. No satisfactory explanation has been found for the wide scatter of the data shortly after the first admission of the gas to the turbine. Other data, however, show that this test condition may be frequently encountered if insufficient time is allowed for equilibrium temperatures to be reached.

The comparative cooling tests for both methods of cooling were made by varying the mass flow of cooling air with all of the other conditions held constant. An additional run was made with the radiation-type cooling cap to study the effect of varying the exhaust-gas temperature. All of the operating conditions were held constant except the fuel-air ratio, which varied the exhaust-gas temperature. The range of operating conditions in these tests is as follows:

Gas temperature at inlet to turbine, $^\circ$F 1240-1440
Gas mass flow, pounds per minute 63.2-86.9
Turbine speed, rpm 10,500-17,000
Turbine-exhaust pressure, inches of mercury absolute . . 8.5-14.85
Turbine pressure ratio 1.55-3.26
Cooling-air flow, pounds per minute:
  Radiation-type cooling cap 0-8
  Convection-type cooling cap 0-11

RESULTS AND DISCUSSION

The turbine-wheel temperatures are plotted against cooling-air flow for the two types of cooling cap with the same turbine operating conditions in figures 6(a) and 6(b). Increasing the cooling-air flow through the radiation-type cooling cap from 0 to 3 pounds per minute reduced the turbine-wheel temperature 20° to 30° F, but additional air flow gave no further cooling of the wheel. (The drop in temperature of the two lower curves for 6 lb/min in fig. 6(b) was due to a sudden change in the temperature of the oil fed to the turbine bearings and was disregarded in fairing the curves.) With the convection-type cooling cap, an increase in air flow from 0 to 3 pounds per minute (average flow for a flight installation) reduced the turbine-wheel temperature by approximately 150° F and further cooling was obtained with additional air flow. Figures 6(c) and 6(d) show that similar results were obtained with different turbine operating conditions. In figure 6(d) for the radiation-type cooling cap, the amount of cooling did not change although the exhaust-gas temperature had been decreased by 150° F, the exhaust-gas mass flow had been increased by 20 pounds per minute, the pressure ratio had been doubled, and the speed had been increased by 6000 rpm. The radiation-type cooling cap obviously has little value as a turbine-wheel cooling device beyond its function of preventing gas flow across the face of the wheel.

Turbine-wheel temperatures are plotted against turbine-inlet gas temperature with all other conditions held constant in figure 7. These temperatures show an approximately linear relation over the

range of gas temperatures tested and, if extended downward, come to a common intersection at -500 on the turbine-inlet gas temperature scale and -400 on the turbine-wheel temperature scale. The following equation may thus be written from which the temperature at any point on the wheel may be corrected for a different turbine-inlet gas temperature if the other turbine operating conditions are the same,

$$T_{wc} = \frac{T_{gc} + 500}{T_{go} + 500} (T_{wo} + 400) - 400 \tag{1}$$

where T is the temperature in degrees Fahrenheit and the subscripts have the following significance:

w   turbine wheel

g   exhaust gas

c   corrected

o   observed

From a number of runs made at different conditions of operation with variable cooling-air flow through the radiation-type cooling cap, three curves were chosen for which all conditions were constant except exhaust-gas mass flow and exhaust-gas temperature. As these gas temperatures fell within the range of figure 7, the turbine-wheel temperatures could be corrected to a common value of 1300° F by equation (1). Figure 8 shows these corrected turbine-wheel temperatures plotted against exhaust-gas mass flow.

SUMMARY OF RESULTS

Tests made of a production turbosupercharger turbine wheel with modified cooling caps over a range of engine and turbine operating conditions showed that:

1. A convection-type cooling cap gave reductions in turbine-wheel temperature up to 150° F with cooling-air flows from 0 to 3 pounds per minute and further reductions with increased air flow.

2. Variations in air flow from 0 to 3 pounds per minute with a radiation-type cooling cap gave maximum reductions in wheel temperature of 20° to 30° F, and no additional cooling was obtained with greater flows.

Aircraft Engine Research Laboratory,
    National Advisory Committee for Aeronautics,
        Cleveland, Ohio, January 31, 1946.

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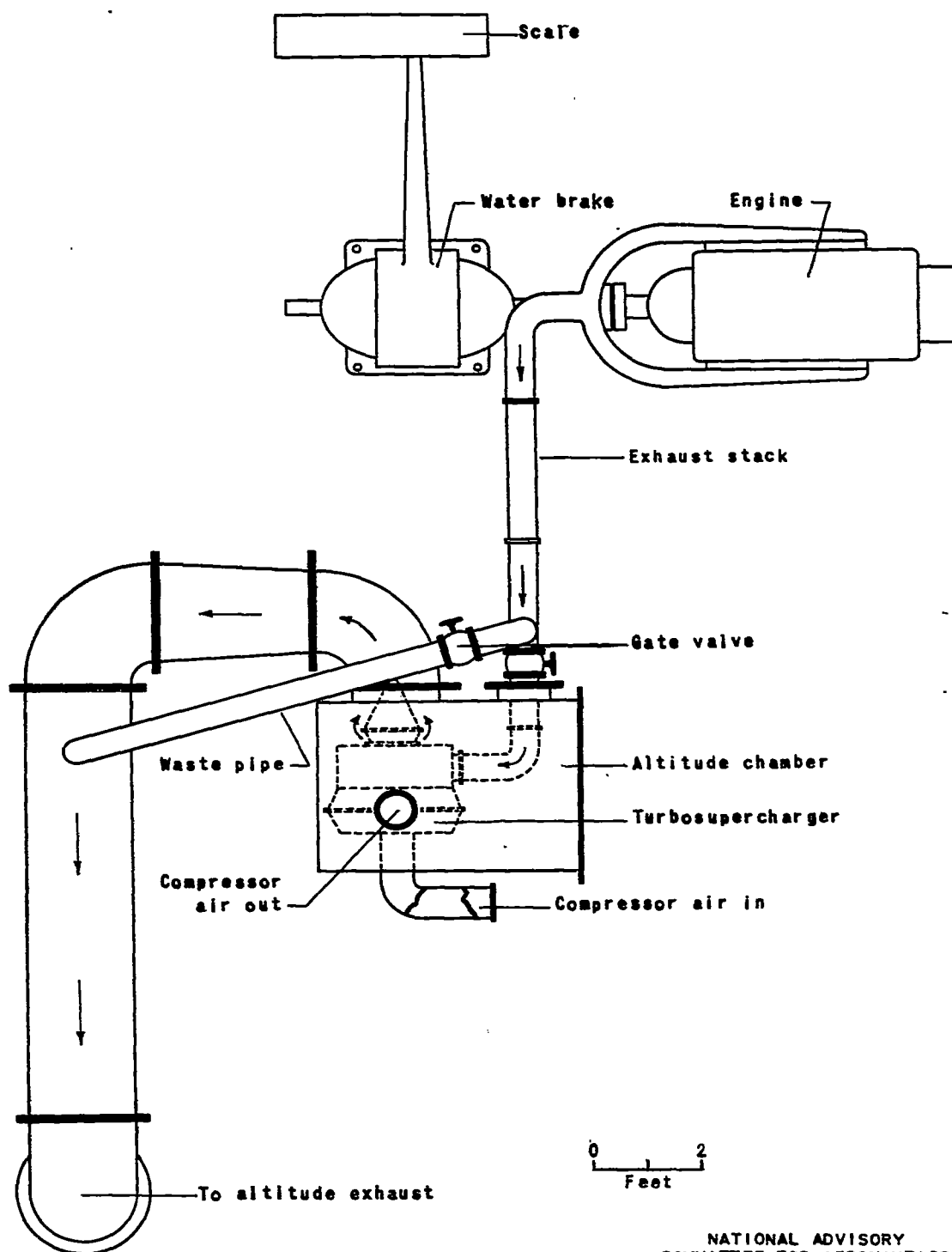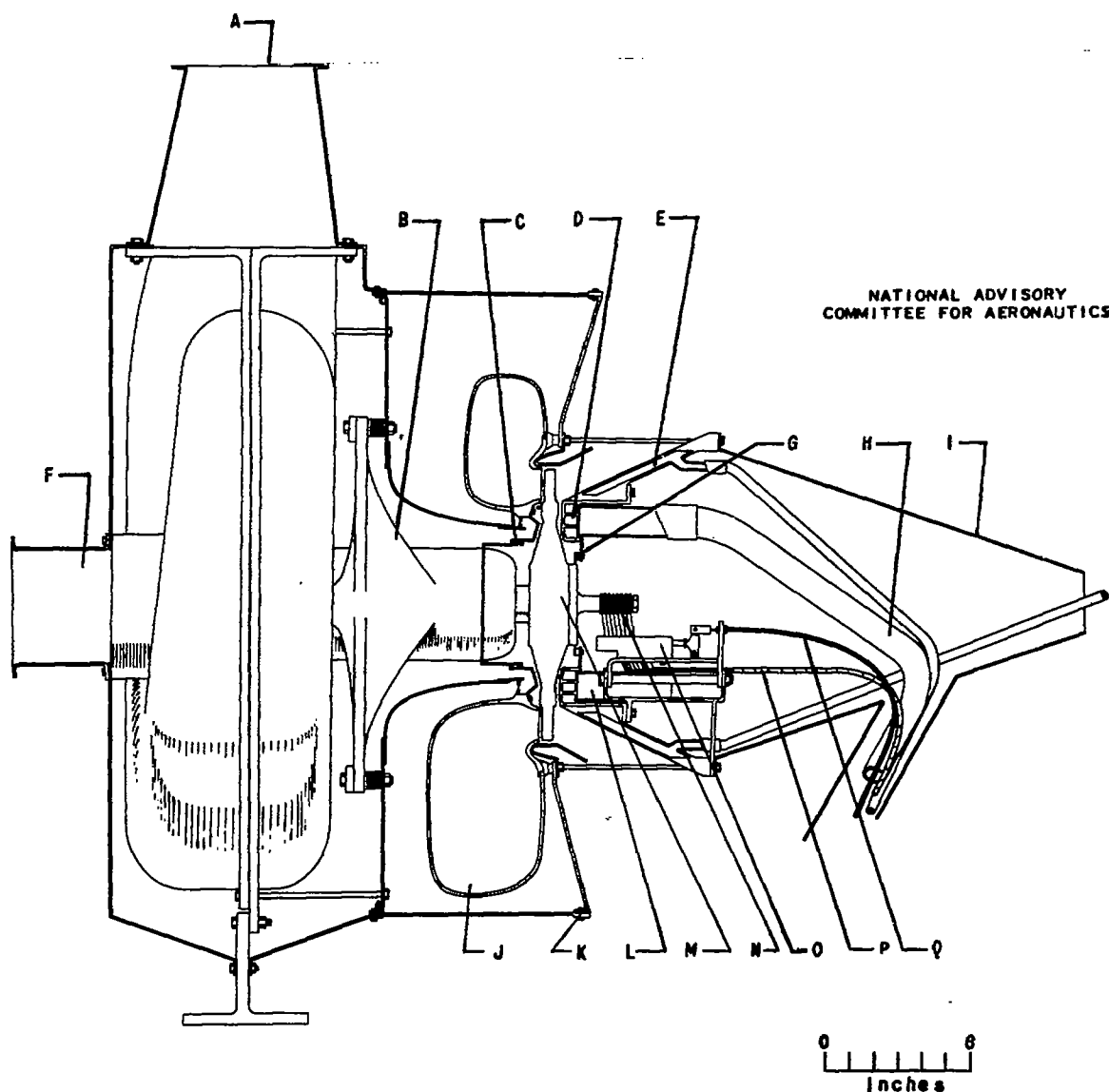
Figure 1. - Schematic diagram of equipment for cooling tests
of turbosupercharger turbine wheel.

Fig. 2   NACA TN No. 1082

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A   Compressor outlet
B   Bearing housing
C   Front seal
D   Cooling cap
E   Water-cooled exhaust hood
F   Compressor inlet
G   Rear seal
H   Cooling-cap air inlet
I   Exhaust-hood cover

J   Nozzle box
K   Nozzle-box cooling-air shroud
L   Cooling-cap air outlet
M   Turbine wheel
N   Slip rings
O   Slip-ring contacts and holder
P   Turbine-wheel thermocouple leads
Q   Slip-ring-contact control cable

Figure 2. - Partly sectioned schematic drawing of turbo-
supercharger used in cooling tests of turbine wheel.

Air-discharge
slots

0   3
Inches

Cooling   Cooling
air in   air out

Air-discharge slots
(in convection cap
only)
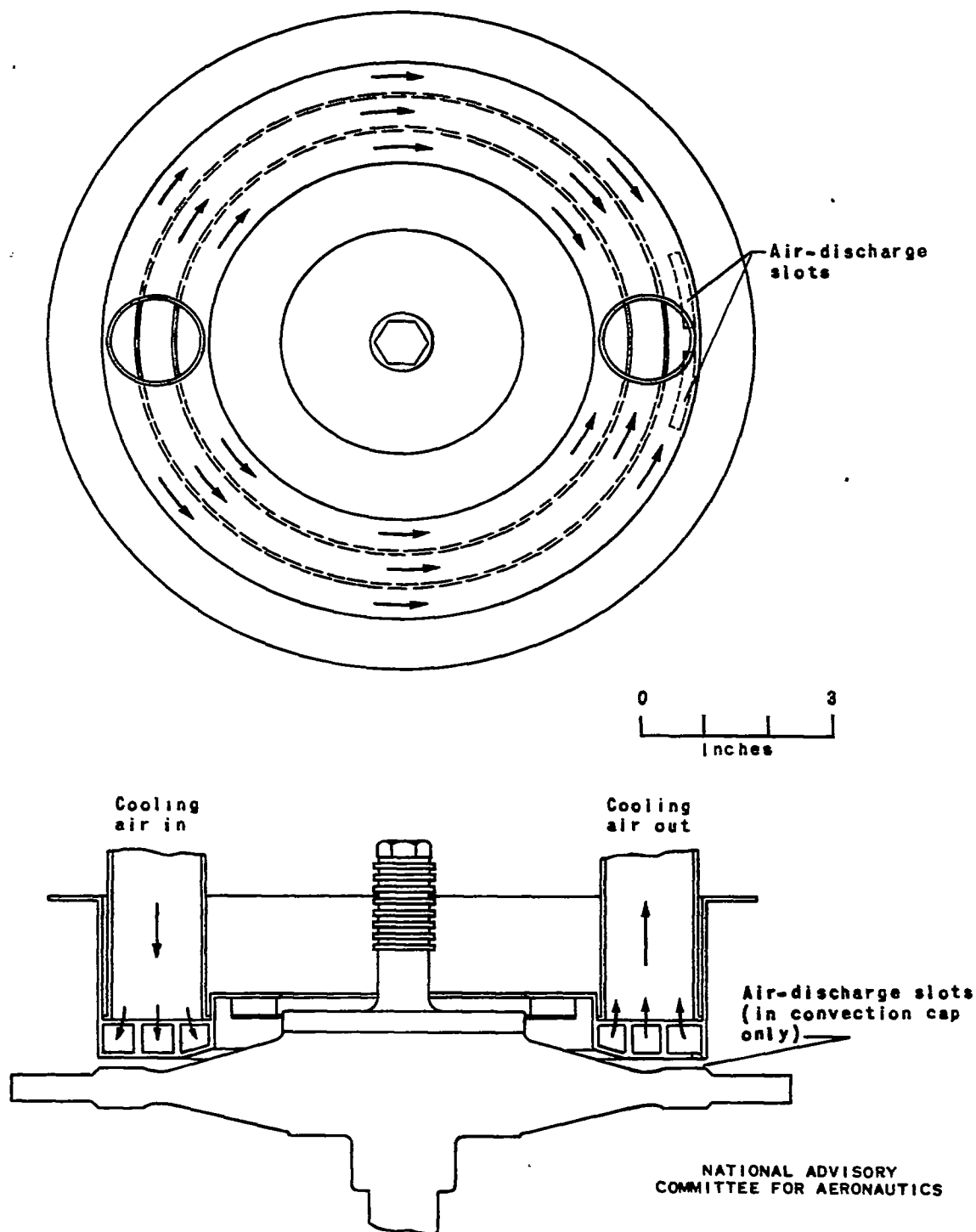
Figure 3. - Modified cooling caps. Radiation-type cap: as
shown but without the air-discharge slots; convection-
type cap: as shown but with the cooling-air exit pipe
blocked off.

Fig. 4   NACA TN No. 1082

0   2

Inches

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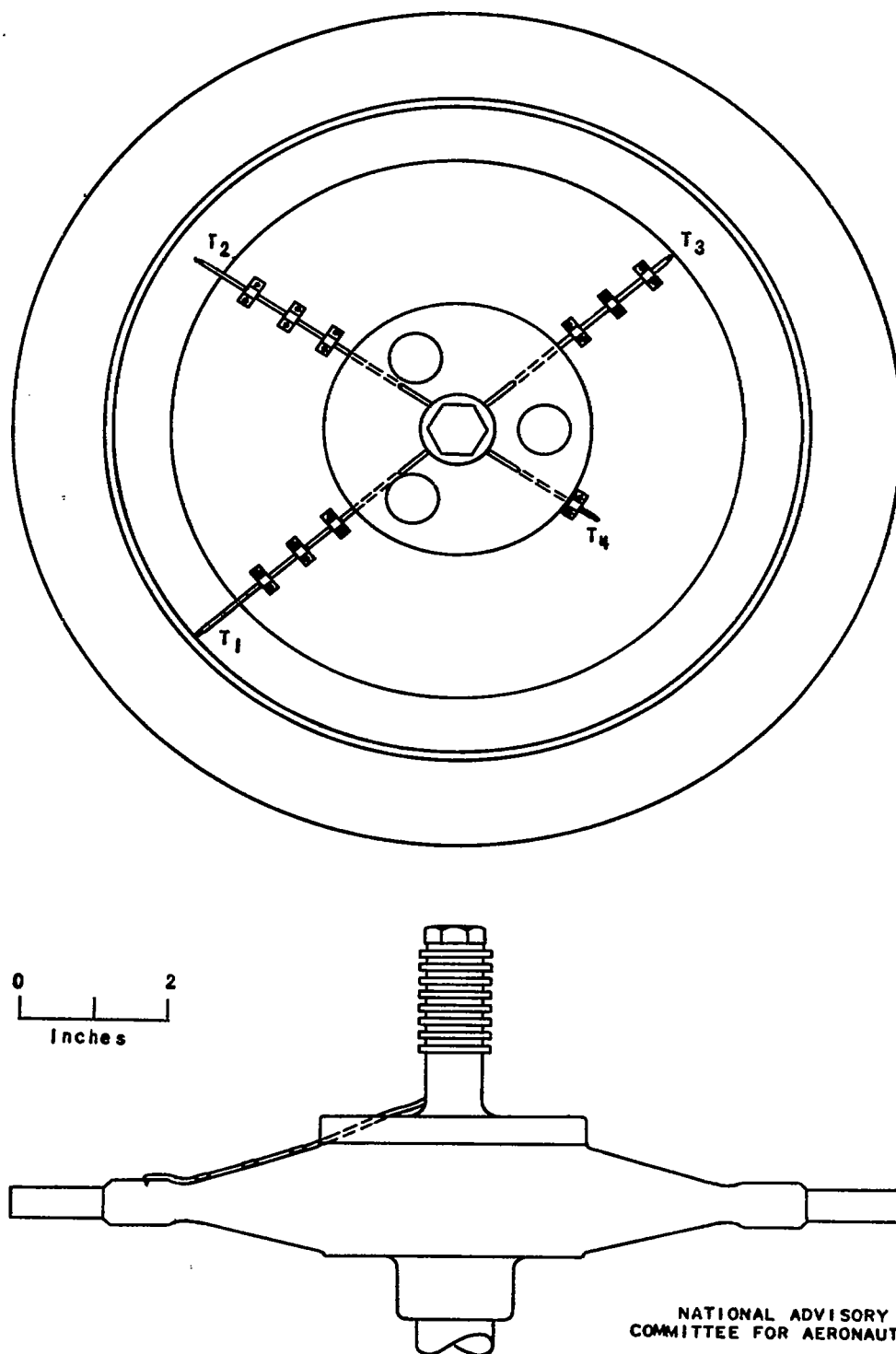
Figure 4. - Location of thermocouples on surface of turbine
wheel.
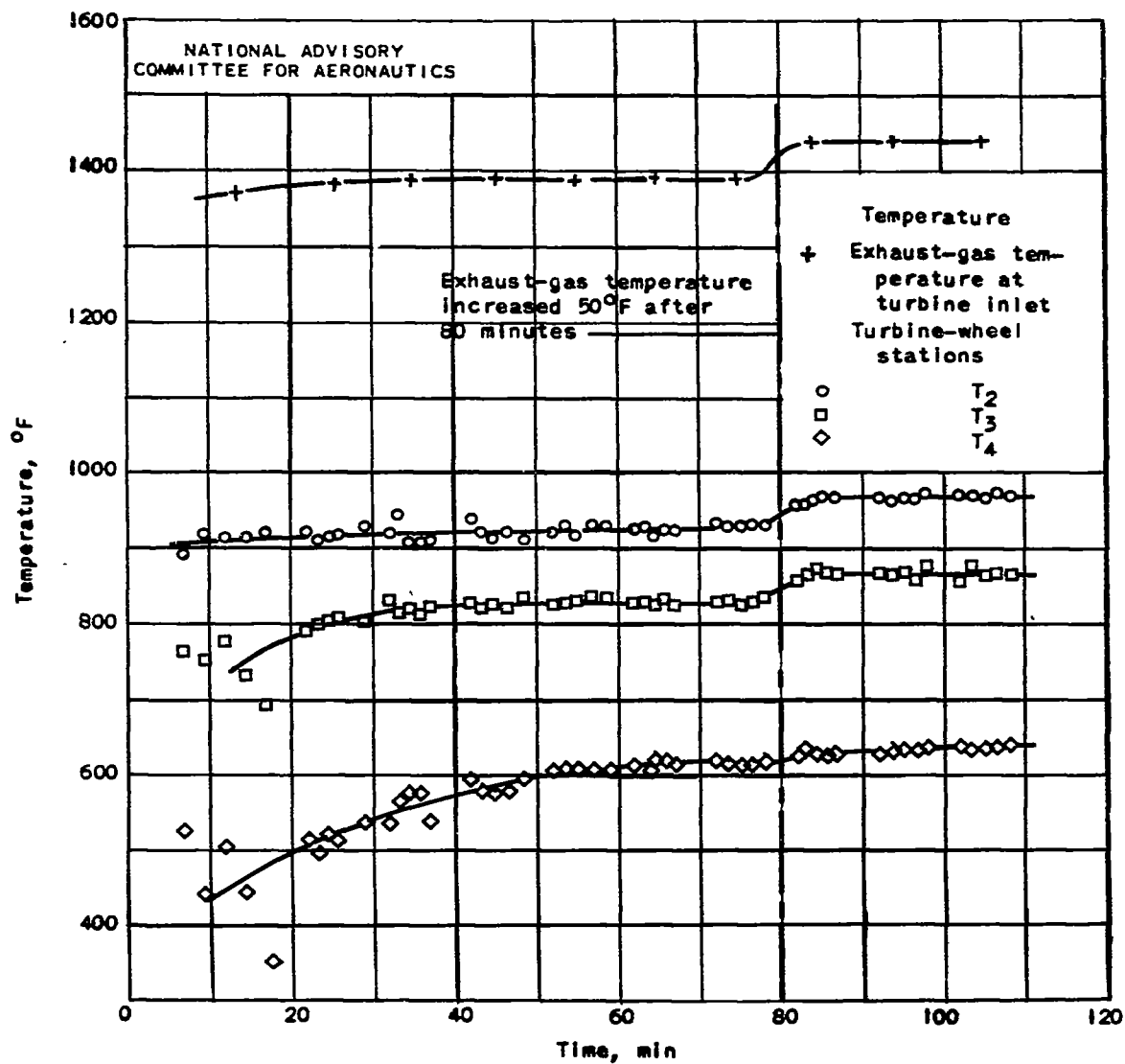
Figure 5. — Variation of exhaust-gas and turbine-wheel temperatures with time. Gas flow, 68.95 pounds per minute; turbine-exhaust pressure, 14.87 inches of mercury absolute; pressure ratio, 1.55; turbine speed, 10,500 rpm; cooling-air flow, 11 pounds per minute.
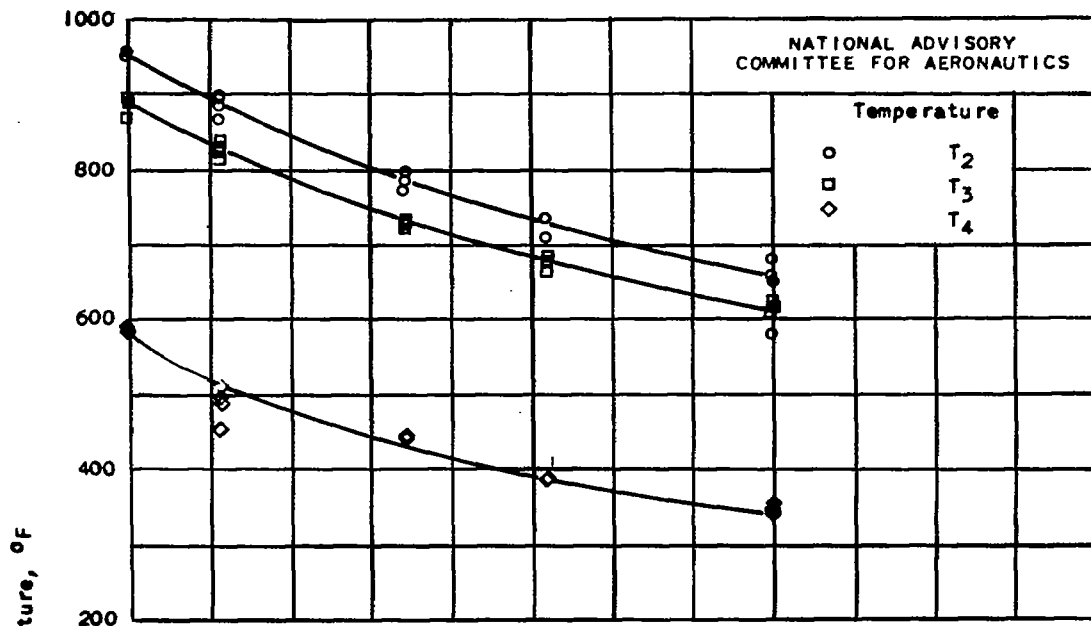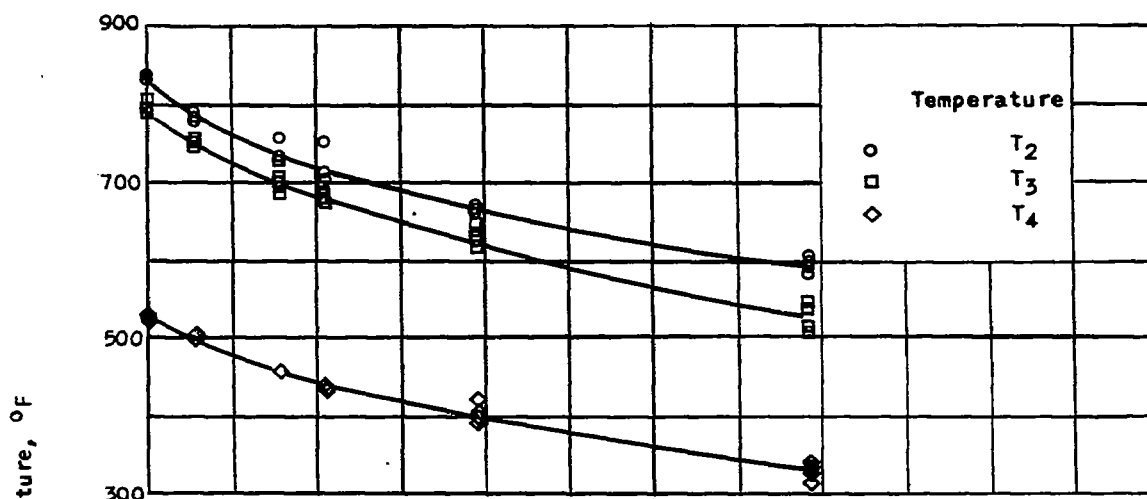
(a) Convection-type cooling cap. Gas flow, 67.4 pounds per minute;
turbine-inlet gas temperature, 1415° F; turbine-exhaust pressure,
14.54 inches of mercury absolute; pressure ratio, 1.59; turbine
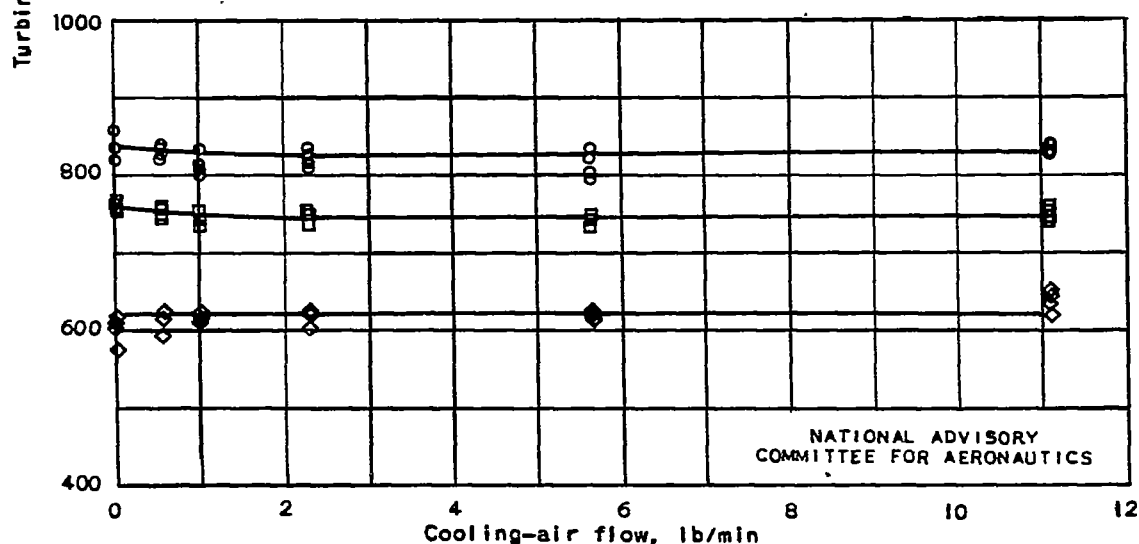speed, 11,300 rpm.
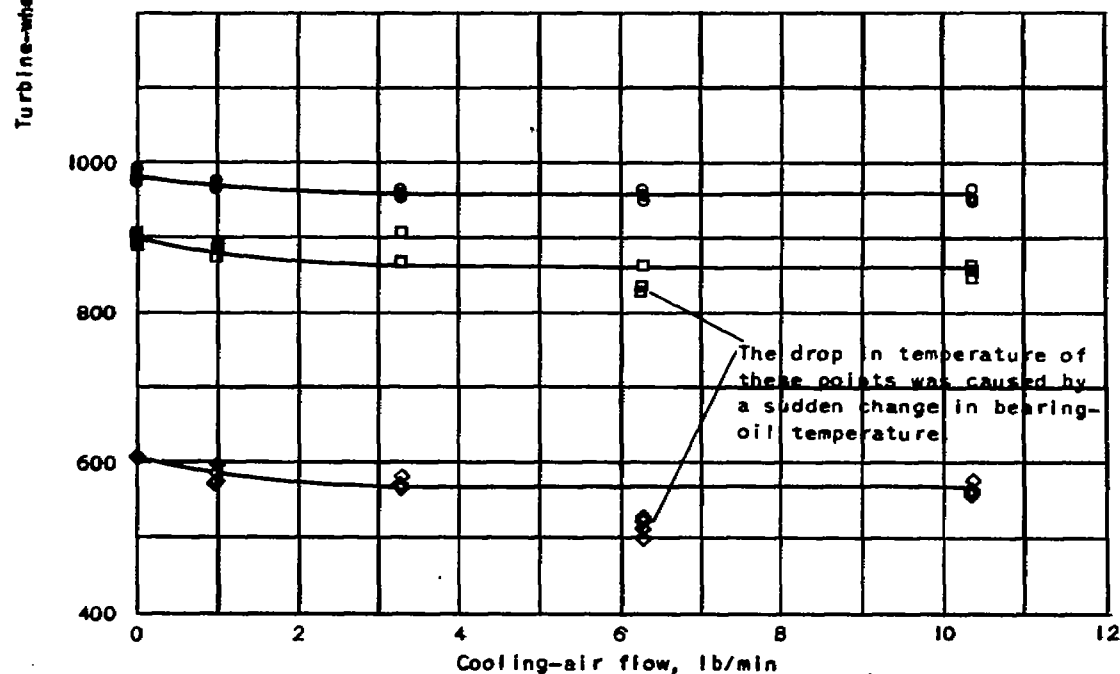
(b) Radiation-type cooling cap. Gas flow, 67.87 pounds per minute;
turbine-inlet gas temperature, 1415° F; turbine-exhaust pressure,
14.56 inches of mercury absolute; pressure ratio, 1.59; turbine
speed, 11,120 rpm.

Figure 6. - Comparison of the cooling results obtained with the convection-type
and the radiation-type cooling cap.

(c) Convection-type cooling cap. Gas flow, 70.8 pounds per minute; turbine-inlet gas temperature, 1253° F; turbine-exhaust pressure, 14.54 inches of mercury absolute; pressure ratio, 1.63; turbine speed, 11,640 rpm.

(d) Radiation-type cooling cap. Gas flow, 86.7 pounds per minute; turbine-inlet gas temperature, 1253° F; turbine-exhaust pressure, 8.57 inches of mercury absolute; pressure ratio, 3.21; turbine speed, 17,000 rpm.

Figure 6. — Concluded.

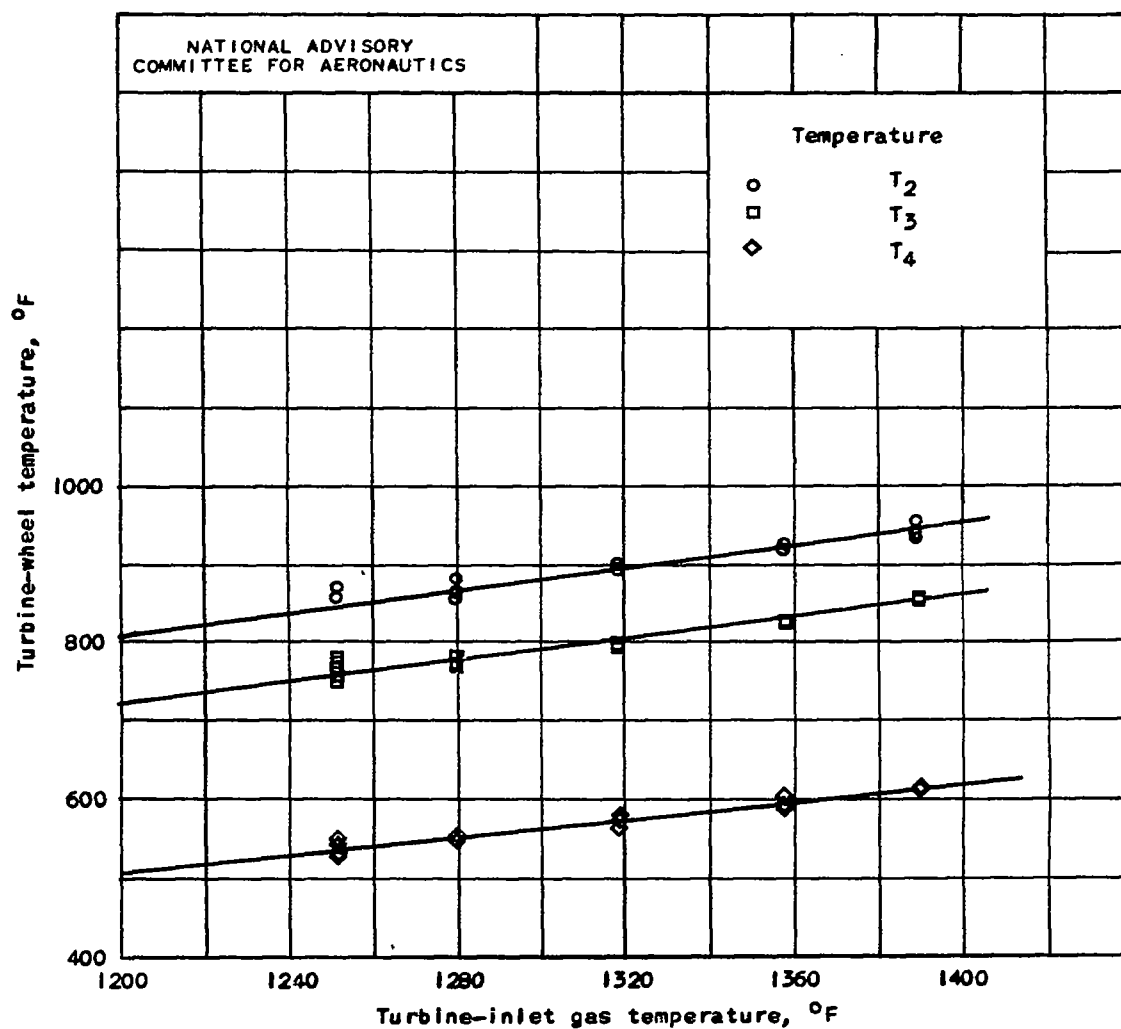Fig. 7   NACA TN No. 1082

Figure 7. — Variation of turbine-wheel temperature with gas temperature using a radiation-type cooling cap. Gas flow, 68.5 pounds per minute; turbine-exhaust pressure, 14.49 inches of mercury absolute; pressure ratio, 1.64; turbine speed, 11,900 rpm; cooling-air flow, 11 pounds per minute.
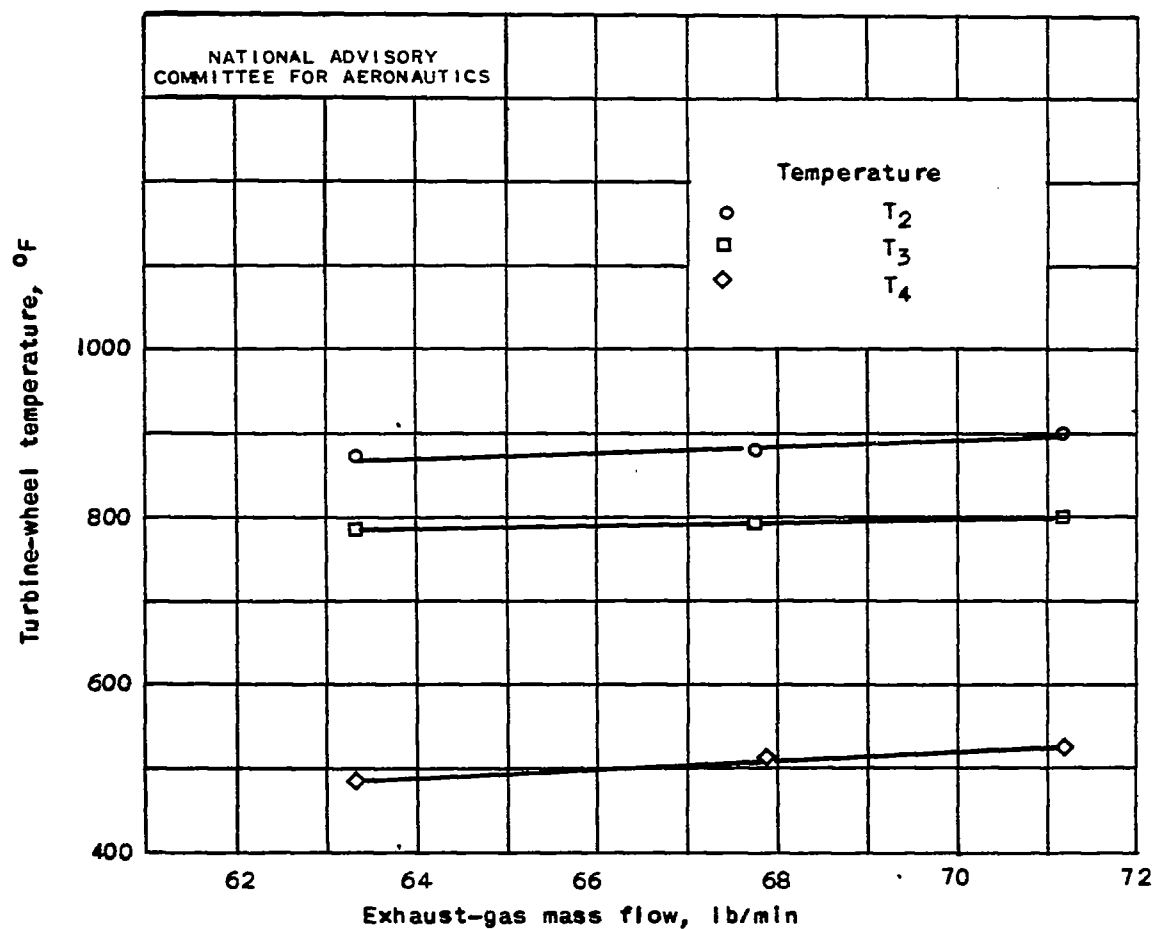
Figure 8. — Variation of turbine-wheel temperatures with exhaust-gas mass
flow using a radiation-type cooling cap. Turbine inlet-gas temperature,
1300° F; turbine exhaust pressure, 14.54 inches of mercury absolute; pres-
sure ratio, 1.61; turbine speed, 11,350 rpm; cooling-air flow, more than 3
pounds per minute.